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PERFORMANCE ANALYSIS OF CAPACITY CONTROL DEVICES FOR HEAT PUMP RECIPROCATING COMPRESSORS

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ABSTRACT

The present paper is concerned with the study of the application of capacity control devices on heat pump reciprocating compressors. Five typical control systems have been studied: variable speed, variable clearance volume, by-pass of the discharge gas, throttling of the suction gas and suction valve cut-off. A simulation model was developed for the analysis.

INTRODUCTION

Capacity modulation of reciprocating compressors has been the subject of intensive development in the last two decades. The adjustment of compressor throughput, at low cost and reliably, to the varying demand of plants in the process and manufacturing industries has now become a necessity. One particular application of reciprocating compressors concerns refrigeration and heat pump systems. The heat pump can be regarded as a variation of the refrigeration plant, with interest being centered on the delivery of heat on the condenser, rather than in the extraction of heat so as to sustain sub-atmospheric temperatures at the evaporator. Nonetheless the basic thermodynamic cycle (vapour-compression) is the same for both systems. A feature of heat pump operation is that the temperature difference between the heat source and heat sink is likely to be more variable than in refrigeration. This results in a wider range of inlet and delivery pressures for the compressor. The thermal load , from the heat source, is also likely to vary considerably [1]. As a result, compressor capacity modulation becomes almost an essential part of heat pump operation. The objective of the present paper is to study the application of capacity control devices on heat pump reciprocating compressors.

There is available a great variety of compressor capacity control devices and a number of surveys can be found in the literature [2,3]. According to Verma [3] capacity modulation can be divided in three basic groups: stepped, stepless and composite control. This is only one criterion as they may be grouped in terms of internal or external control [2], or in regard to the point of application (suction/discharge valve, cylinder volume or drive actuating control). In addition to the traditional start-stop regulation the following devices are studied in the literature: compressor uncoupling, variable speed [4], variable clearance volume [5,6], variable cylnder volume [7], suction valve cut-off [8], suction valve unloading [9,10,11], blocked suction [12], top head unloading (internal gas by-passing) [13,14], suction throttling [15], discharge gas by-passing [15]. The list is not complete as there is a number of variations and combinations of the basic devices [2].

In the present paper a numerical simulation model was employed to analyse five devices: variable speed, variable clearance volume, by-pass of the discharge gas, throttling of the suction gas and suction valve cut-off. To find the most suitable device several aspects should be taken into account. They would include the energy conversion efficiency, type of application, cost and reliability, to name but a few. Therefore, it should be stressed that the objective of this paper is restricted to the thermodynamic performance of the devices. A similar analysis has been carried out by Haseltine and Qvale [16] for three different capacity reduction methods. Results were limited by the simplifications made on the compressor model. As far as capacity control in heat pumps is concerned, the majority of the papers [17,18,19] concentrate on variable compressor speed.

SIMULATION MODEL

The heat pump under consideration is presented schematically in Figure 1. It is of the vapour-compression type and consists of an open-type reciprocating compressor, a water-cooled condenser, evaporator and expansion valve. For the purposes of the present study, it is assumed that the evaporator operates at constant pressure, supplying vapour at a constant degree of superheat. This assumption limits the simulation to the compressor and condenser unit. Therefore, vapour conditions at the compressor inlet are supplied to the model as input data. In addition to have a less complex model, the reason for this simplification was that it allowed the study of the sole effect of compressor capacity modulation on the condenser water outlet temperature.

A detailed model for the compressor was utilized. It follows traditional methods of computer simulation for reciprocating compressors, by considering the cylinder space as a control volume, Figure 1, with two flow boundaries (valves), a moving boundary (piston) and heat transfer across its surface (dQ/dt). The energy equation, in its time-derivative form, is applied to the control volume, leading to,

$$\frac{dP}{dt} = -\frac{1}{V}\left[\frac{dQ}{dt} - m\frac{dh}{dt} - (h - h_i)\frac{dm}{dt}\right] \tag{1}$$

where P and V are the instantaneous cylinder pressure and volume and m and h, the gas specific enthalpy and mass. Integration of equation (1) gives the variation of cylinder pressure with time. Further equations are required for the determination of the right-hand side terms of (1). They are: the heat transfer equation (dQ/dt), kinematics equation (V), equation of state (m), enthalpy equation (h) and valve mass flow rate equation (dm/dt). Real gas equations [21] are used for the evaluation of m and h. Pressure fluctuations in the inlet and delivery ducts are ignored, so that evaporating and condensing pressures are considered to be steady throughout the cycle. For the mass flow rate through the valves, one-dimensional flow is assumed. The model also requires a detailed information on the valve system, which includes: valve and spring masses, spring stiffness, valve pre-load, viscous damping factors, drag and discharge coefficients, maximum displacement and flow areas. One original aspect of the model [20] concerns the generalized equation for the effective flow area, A_f, as related to valve lift, y.

$$A_f = C_d\, A_{max} \sin\left[\frac{\pi}{2}\left[\frac{y}{y_{max}}\right]\right] \tag{2}$$

Equation (2) was found to agree well with experimental data from several authors, Figure 3. Valve lift and pressure difference are related by the dynamics of valve motion (one-degree of freedom model). In addition, energy and mass conservation equations are applied to account for the presence of suction throtling or gas by-passing. There is, of course, no need to model variable speed or variable clearance volume, as this can be done simply by altering the input data. Modelling the suction valve cut-off mechanism is done by imposing an instantaneous closure of the valve (ie, zero displacement). The condenser is modelled by the usual three heat exchanger equations (heat balances in water and refrigerant streams and the log-mean temperature difference equation).

The model basic algorithm is presented below:

a) With suction conditions (T and P) and an estimated discharge pressure perform the compressor model, in order to provide the refrigerant mass flow rate and the discharge temperature.

b) From condenser inlet conditions determine the condensing and water outlet temperatures and the thermal power output.

c) Compare the estimated condensing pressure with that used for the compressor model. If convergence in P_{cd} has not been achieved return to (a) with a new value.

RESULTS AND DISCUSSION

The model was applied to a typical medium-sized heat pump system. A two-cylinder compressor operated at a nominal speed of 1500 rpm, with suction at 10^{o}C and 3 bar. Clearance ratio was set at 0.0363 with a bore/stroke relation of 0.0667/0.0635 m. The condenser was assumed to have an overall heat transfer coefficient of 0.4 kW/^{o}C and 2

m^2 of heat transfer area. Water inlet temperature was kept at $30^{\circ}C$. Refrigerant-12 was taken as the working fluid.

In a first comparative analysis, Figure 5, the compressor was supposed to run at a constant pressure ratio of 5, with the mass flow rate ranging from 100% to 30% (mass flow ratio of 0.3) of the design point. Figure 5a depicts the variation of the volumetric efficiency with the mass flow ratio. It can be seen that three of the capacity control devices (suction throttling, variable clearance volume and suction valve cut-off) do act on the compressor volumetric efficiency. This is not the case for the gas by-passing, as the efficiency is defined in terms of the mass flow rate at the discharge port (d), Figure 4a. As for the variable speed device, lower shaft speeds resulted in reduced gas flow losses, leading to an increase in the volumetric efficiency.

With the exception of gas by-passing, where practically the same quantity of gas must always circulate through the compressor, all devices presented a reduction on power consumption for lower mass flow ratios, Figure 5b. However, this situation changes when the comparison is made in terms of the specific power consumption (Watts per kg/s of gas), Figure 5c. Variable clearance volume and variable speed stand out as the most effective methods, with suction throttling and suction valve cut-off presenting the worst results. Such behaviour will directly affect the heat pump performance.

One important parameter for heat pump applications refers to the compressor discharge temperature, which must not exceed the refrigerant thermal stability limit. Again, Figure 5d, variable clearance volume and variable speed presented, by far, the best results. In fact, the other devices presented a serious tendency of increasing the discharge temperature at reduced mass flow ratios. Without a proper design this could jeopardize the entire control effort.

In a second analysis the model was employed to simulate the operation of the heat pump with three different water flow rates (0.154, 0.24 and 0.49 kg/s), each providing, at the design point, an outlet temperature of $60^{\circ}C$, $50^{\circ}C$ and $40^{\circ}C$, respectively. From the predicted results, Figure 6, it can be concluded that, in theory, all five devices were able to provide a wide range of condenser outlet temperatures. Water inlet temperature, for all cases of Figure 6, was $30^{\circ}C$.

Figures 7 and 8 summarize the heat pump analysis, showing the variation of the heating coefficient of performance and the compressor discharge temperature with the condenser water outlet temperature. Clearly, variable clearance volume and variable speed presented the best result: water temperature reduction with an increasing COP and within safe limits for the refrigerant stability. In fact, results from Figure 7 have been anticipated by Figure 5c, with the difference that the compressor pressure ratio, dependant on the condensing pressure, is now varying. Not surprinsingly, the devices that make use of irreversible processes, such as throttling (ST,BP) and gas mixing (BP), presented the worst performances, not only in respect to the COP but also regarding the temperature levels achieved at the discharge. As for the suction valve cut-off method, COP figures were not adversely affected, eventhough Figure 8 shows that the discharge temperature may be cause for concern.

CONCLUDING REMARKS

The main objective of the paper was to assess how the condenser water outlet temperature could be controlled by means of varying the compressor capacity. For that a simulation model for the heat pump was developed. It included the model for the compressor itself (with constant intake and delivery pressures, valve dynamics and heat transfer across cylinder walls) as well as a simplified model for the condenser.

To compare all five mechanisms two parameters have been selected: the heat pump coefficient of performance and the compressor discharge temperature, this one being important in the refrigerant thermal stability aspect. Generally it has been concluded that best results were obtained with variable speed and variable clearance volume. Suction throttling and discharge gas by-passing, being highly irreversible processes, presented the worst performances.

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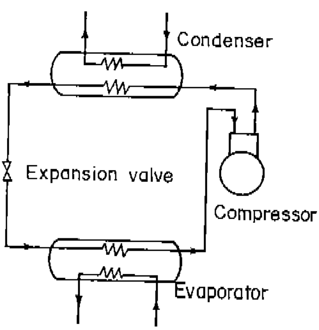


Fig. 1 Vapour-Compression Heat Pump System.

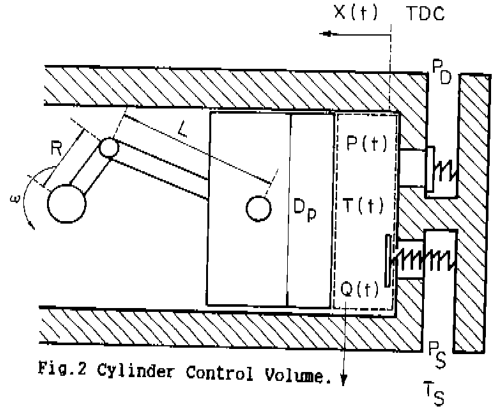


Fig.2 Cylinder Control Volume.

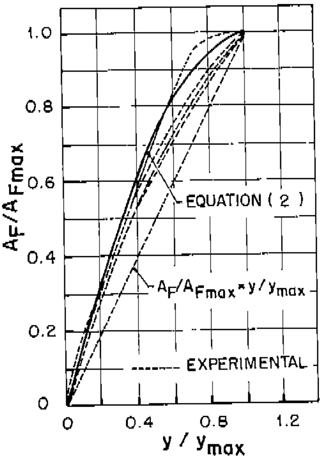


Fig.3 Valve Effective Flow Area vs. Valve Displacement

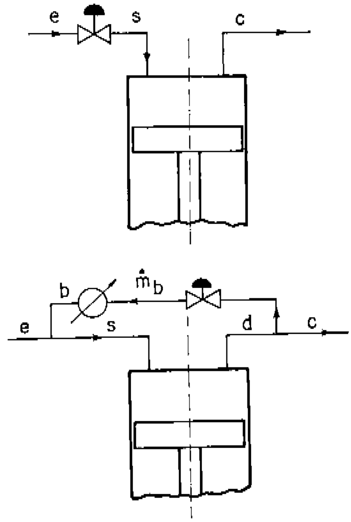


Fig.4 (a) Suction Throttling; (b) Gas By-Passing.

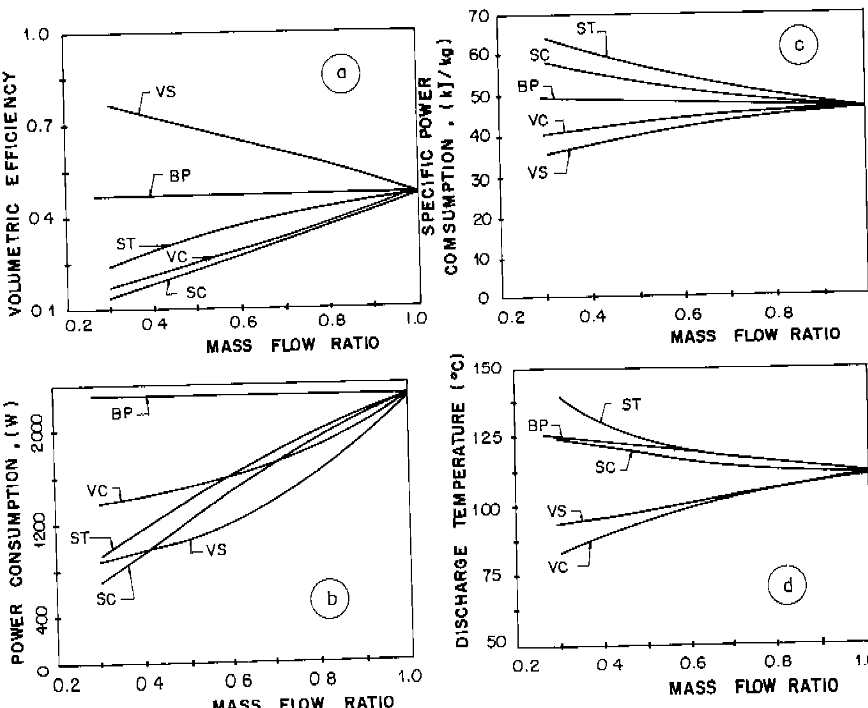
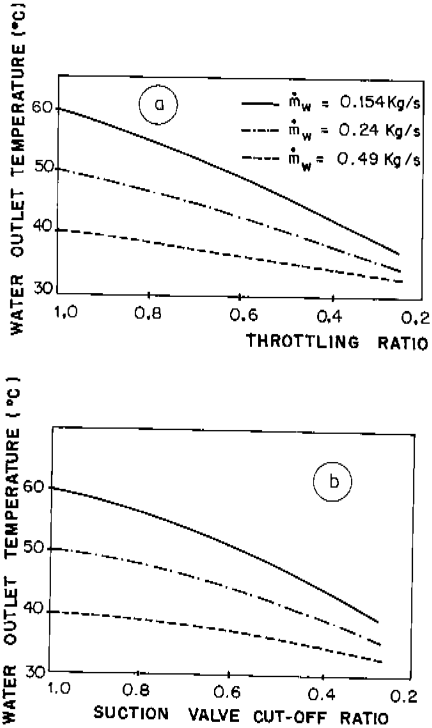


Fig.5 Effect of Capacity Control on: a) Volumetric Efficiency; b) Power Consumption; c) Specific Power Consumption; d) Discharge Temperature. ST: Suction Throttling; SC: Suction Valve Cut-off; BP: Discharge Gas By-Passing; VC: Variable Clearance Volume; VS: Variable Speed.



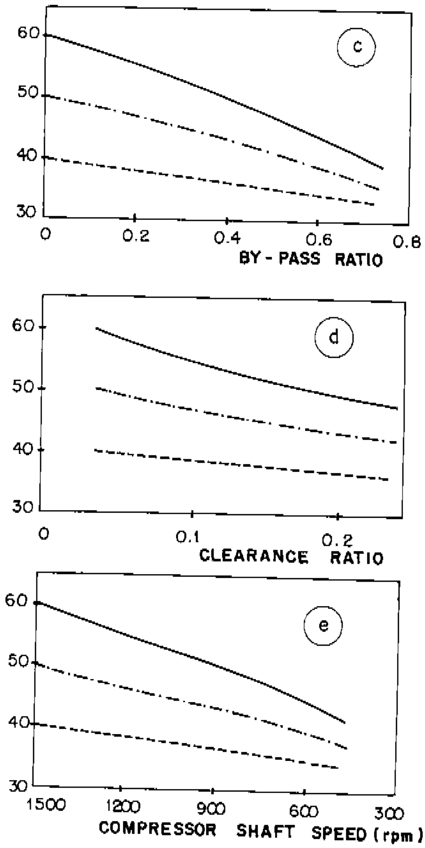


Fig.6 Control of Hot Water Temperature with: a) Suction Throttling; b) Suction Valve Cut-off; c) Discharge Gas By-Pass; d) Variable Clearance Volume; e) Variable Shaft Speed.

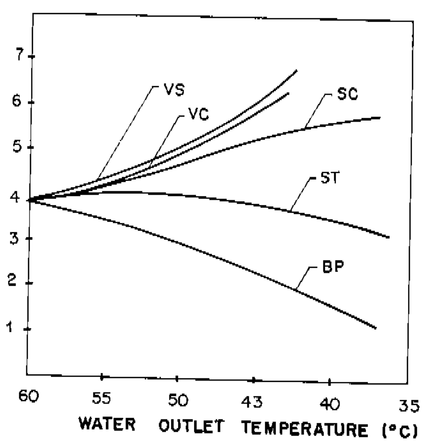


Fig.7 Heat Pump COP vs. Condenser Water Outlet Temperature

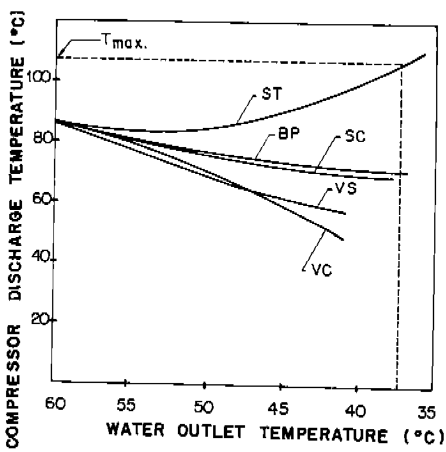


Fig.8 Compressor Discharge Temperature vs. Water Outlet Temperature